(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 541 771 B1

(12) EUROPEAN PATENT SPECIFICATION

(45) Date of publication and mention of the grant of the patent: **24.07.1996 Bulletin 1996/30**

(21) Application number: **92912302.4**

(22) Date of filing: **26.05.1992**

(51) Int Cl.6: **F16H 37/08**

(86) International application number: **PCT/ES92/00046**

(87) International publication number: **WO 92/21896 (10.12.1992 Gazette 1992/31)**

(54) **Continuously variable transmission**

Stufenlos arbeitendes mechanisches Getriebe

Transmission à réglage continu

(84) Designated Contracting States: **AT BE CH DE DK FR GB IT LI LU NL SE**

(30) Priority: **31.05.1991 ES 9101327**

(43) Date of publication of application: **19.05.1993 Bulletin 1993/20**

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Printed by Jouve, 75001 PARIS (FR)

Description

The present invention relates to a stepless transmission, specially intended for motor vehicles and it is designed in such a way, as to allow a considerable increase in the amount of power to be transmitted with respect to conventional stepless transmissions.

At present, some stepless transimission mechanisms are known, WHICH basically consist of pair of pulleys mounted on parallel shafts and the movement transmission between them is obtained by means of a trapezial cross section belt. Each one of these pulleys is constituted by two interlocked coaxial trunco-conical plates or pieces, WITH their internal face to face shapes are of trunco-conical configuration. The trunco-conical plates and pieces of each pulley can be shifted between them, moving away or approaching, modifying in this manner the annular zones where the belt will be supported. When approaching, the trunco-conical plates or pieces of each pulley, the radii of the annular zones where the belts lie will be increased. On the other hand, when moving away, the trunco-conical plates or pieces of each pulley, the contrary is obtained. These radii variations in the pulleys permit the attainnent of a continuous variation of the transmission ratio.

It is .known by patent US-A-4 470 326 from which are known the features outlined in the preamble of claim 1 a power transmission which has a variable belt drive, a two-speed planetary gear arrangement and a countershaft type gear arrangement disposed in combination to provide a continuously variable synchronously shifting transmission. The planetary gear arrangement and the variable belt drive are disposed in parallel drive relationship thereby reducing the belt load. The planetary gear arrangement has two output members selectively connectible via clutches with respective countershafts in the countershaft type gear arrangement. The countershafts support selectively engegeable ratio gears which mesh with respective drive gears on the transmission output shaft in a manner to provide pairs of identical gear ratios between the shafts such that ratios on each countershaft may be simultaneously engaged and the output clutches of the planetary gear set can be interchanged at a synchronous speed point to permit a ratio change. Speed change within any given gear ratio is provided by controlling the ratio of the belt drive. In one gear ratio, the output speed of the belt drive is increased to provide an increased transmission speed while in the next succeeding ratio, the output speed of the belt drive is decreased to provide an increase in the transmission output speed. The transmission is usable with a constant speed prime mover to provide a continuously variable speed range for a vehicle which is greater than the continuously variable speed range for any given speed ratio.

It is interesting to outline that a metallic "belt", nowadays used, has overcome the main problem of the formerly used "belt", made of rubber or elastomeric materials with or without reinforcement trends, which is the low value of the peak power that could be transmitted. With the metallic "belt", this problem has been overcome because it works under compression conditions instead of traction conditions.

The above-mentioned metallic "belts" and the trunco-conical pulleys have provided an important improvement in the amount of the maximum power to be transmitted, in a continuous form in its speed variation, allowing an special application in the automotive field. In spite of this, the amount of maximum power that has been transmitted is relatively low, so that these stepless transmissions can only be used on motor vehicles with low power engines.

The object of this invention is to dispose a stepless transmission, of mechanical constitution, that allows a substantial increase, in the amount of maximum power that could be transmitted, with the undoubted advantages that represents in the automotive field.

On the one hand, it will allow to develop a stepless transmission to be fitted on motor vehicles with medium and low power engines, with a lower cost with respect to the present models, since they can be equipped with conventional "belts" traction working.

On the other hand, the stepless transmission application could be extended to passenger cars with high power engines, and in the same way to the commercial vehicles, such as trucks, buses, etc.

The present stepless transmission invention consists of traditional mechanical elements such as, "belt" and trunco-conical pulleys transmissions, gear wheel sets, e.g., epicyclic gears or planetary gears, synchro sleeves for gear selection, brakes and/or multidisc clutches, etc. and it is based on the fact that through the trunco-conical pulleys and the corresponding "belts", only is transmitted a fraction of the total power supplied by the vehicle engine.

The purpose of this invention is to provide a transmission arrangement including a forward drive and a reverse drive, a V-belt-type continuously variable transmission with at least a pair of pulleys connected by a flexible transmission element, a power transmission mechanism and an output planetary gear train composed of a carrier for a plurality of planet gears, a sun gear and a ring gear, whereby one pulley of said pulleys is connected to a transmission input shaft receiving power from an engine and the other pulley is drivingly connected to an input shaft of said power transmission mechanism which has an output gear drivingly connected to said sun gear of said output planetary gear train, whereby one element of said ring gear or said carrier of said output planetary gear train is drivingly connected to said transmission input shaft and the other element is connected to an output shaft of the transmission arrangement which shaft being an input shaft of a stepped conventional gearbox, characterized in that said power transmission mechanism is provided with a forward/reverse drive shift mechanism so that upon its actuation

said sun gear of said output planetary gear train rotates either in one or in the opposite direction.

The forward/reverse drive shift mechanism can be constituted by a clutch and a brake that respectively act for the said gears.

According to another variant the forward/reverse drive shift mechanism consists of a displaceable sleeve which, displaced in one direction engages the toothed wheels, and in the opposite direction brakes the toothed wheel when acting on a fixed element.

According to a third solution the forward/reverse drive shift mechanism consists of a sleeve capable of optionally engaging the wheel that is integral with the shaft of the pulley leading to one gear train or to another.

The features and advantages of the present invention, just as they are stated in the claims, will now be described in detail with reference to the appended drawings showing several examples of the stepless transmission achievements of the present invention and their possible applications.

Figure 1 is a schematic section of the stepless transmission basic element of the invention, which henceforward will be designated as "Module of continuity and power deviation".

Figures 2 to 6, are similar views to Figure 1 showing possible variations of the "Module of continuity and power deviation".

Figures 7 to 20, correspond to schemes of different epicyclic gear sets and trunco-conical pulleys for power transmission by means of a "belt" that can be used in the present stepless transmission invention.

Figures 21 and 22 show graphically the " Module of continuity and power deviation " operation.

Figures 23, 25 and 26 are schematic sections of achievements examples of stepless transmissions in accordance with the invention.

Figure 24 corresponds to the kinematic operation diagram of the stepless transmission shown in Figure 23, fitted on a passenger car with enough powered engine to reach a maximum speed of 210 km/h.

Figures 27 and 28 indicate two possible achievement variations on the " Module of continuity and power deviation ".

Figure 1 shows the " Module of continuity and power deviation ", the references numeral 1 and 2 indicate the variable transmission ratio pulleys, constituted by trunco-conical plates or pieces with a common shaft 3, with trunco-conical surfaces face to face to supply support to the "belt" 4. The reference numeral 5 indicates a gear wheel which is coaxial and interlocked to the gear crown 7 of an intermediate epyciclic gear 11. The reference numeral 6 indicates the satellites of the planetary gear set or epicyclic gear 11. The reference numeral 8 is a clutch which, when operating, interlocks the crown gear 7 to the satellites carrier of the epicyclic gear 11. The "Module of continuity and power deviation " also includes a brake, reference numeral 9 which, when operating, immobilizes the satellites carrier of the epicyclic gear 11. In this gear, the reference numeral 10 indicates the sun which is coaxial and interlocked to the shaft 3. The references numeral 12 and 14 indicate the crown gear and the satellites of an output planetary gear, generally reference numeral 16. The satellites carrier of this second gear, has the reference numeral 13. The reference numeral 15 indicates a gear wheel that meshes an intermediate gear wheel 17, which, in turn, meshes the gear wheel 5. Letter E indicates the input shaft, whereas letter S indicates the output shaft.

Figure 2 corresponds to a similar achievement to the one described in Figure 1, except that the gear wheel 5 is coaxial and interlocked to the shaft 3. Moreover, the sun 10 is coaxial and interlocked to the gear wheel 15.

The clutch 8 operates, in any case, between any two elements of the planetary gear, since it is well known, produces the same effect, that is to make equal the rotation speed of the three elements: the sun, the crown gear and the satellites carrier.

In Figure 1, the two epicyclic gears shafts are parallel, whereas in Figure 2 they are coaxial. In both achievements, the input and output shafts are coaxial.

Figure 3 shows the same arrangement to the one shown in Figure 1. The same references are used in Figure 3 as in the following Figures to indicate the same elements or components. The only difference between Figure 3 with regard to Figure 1 is that the input shaft E is placed on the shaft left side of the pulley 1.

Figure 4 shows the same achievement to the one shown in Figure 2. The only difference between Figure 4 with regard to Figure 2 is that the input shaft E is placed on the left side of the pulley 1. In Figures 3 and 4, the reference numeral 18 indicates a housing to be fastened to the engine flywheel housing.

Figure 5 achievement, is similar to the one shown in Figure 2, but is different in that the gear wheel train that meshes between them, reference numeral 5, 15 and 17 have been deleted and another pair of pulleys, reference numeral 19 and 20, have been added. The first of these pulleys, reference numeral 19, is equal to the reference numeral 1, whereas the pulley 20 is equal to the reference numeral 2. The rest, its constitution, operation and references correspond to the explanations exposed with respect to Figure 2.

Figure 6 achievement, is similar to the one shown in Figure 4, but is different in that the gear wheel train that meshes between them, reference numeral 5, 15 and 17, have been deleted and in the same manner as in Figure 5, another two pulleys, reference numeral 19 and 20, have been added.

Figures 7, 8 and 9 represent the same set of the two epicyclic gears already described and with the reference numeral 11 and 16. The references numeral 6, 7 and 10 indicate, respectively, the satellites, the crown gear and the sun of the gear wheel 11, whereas the references numeral 14, 12 and 21 indicate, respectively, the satellites, the crown gear and the sun of the planetary gear 16. The reference numeral 22 indicates a training gear

wheel, which is coaxial and interlocked to the sun 10. The reference numeral 23 indicates a training gear wheel, which is coaxial and interlocked to the satellites carrier 26 of the planetary gear 11. The reference numeral 24 indicates a inside indented sleeve and the reference numeral 25 indicates a fixed gear wheel. The sleeve 24 includes a synchronization mechanism to interlock the gear wheels 22 and 23 on its shift to the left or to interlock the gear wheels 23 and 25 on its shift to the right.

In Figure 8, the sleeve 24, when shifted to the left, interlocks the gear wheels 22 and 23. In Figure 9, the references numeral 8 and 9 indicate, respectively, a clutch and a brake, as already indicated for Figures 1 to 6.

Figures 10 to 20, correspond to different sets of pulleys, gear wheels and epicyclic gears. The reference numerals indicated in these Figures have the meanings already explained for the above Figures. The reference numerals 27 and 28 of Figure 10 indicate a pair of gear wheels which mesh between them. In the different Figures, the references numerals 29 and 30 indicate coaxial shafts, the second of which is of tubular configuration. Also the reference numeral 31 indicates a hollow shaft of tubular configuration.

The combination of one of the two Figures 10 or 11 with one of the Figures 11, 13, 14 and, finally, with one of the Figures 15 to 20, permits to dispose of a group of sets that constitutes, besides the group of sets already explained in the above Figures, a " Module of continuity and power deviation " as will be explained later.

The diagram of Figure 21 shows the amount of the maximum power transmitted through the pulleys of the" Module of continuity and power deviation", measured in percentage of the power supplied by the motor vehicle engine and that corresponds to the Y axis D, of this diagram, in function of the actuation field, C, which values are indicated in the X axis. C is the quotient between the maximum and the minimum values of the transmission ratio.

The diagram of Figure 22 includes two graphics with a common X axis, which are represented by the K values (transmission ratio of a pair of trunco-conical pulleys, i. e. the quotient between the arcs radii that the "belt" describes in each pulley). The lower diagram shows the power values D, as explained for the Figure 21, whereas the upper diagram shows in the Y-axis the values T that is the transmisson ratio of the " Module of continuity and power deviation ". These diagrams have been prepared for prefixed values of parameters that define the epicyclic gears and trunco-conical pulleys, corresponding to one achievement example that later, when explaining the operation, will be commented again.

Figure 23 represents, schematically, one example of the engagement of the " Module of continuity and power deviation " of Figure 1 to a gearbox assembly, with two forward transmission ratios and one for the reverse gear. The set so constituted corresponds, for example, to a stepless transmission for a passenger car equipped with a transverse engine and front wheel traction. As in the above Figures, letters E and S indicate, respectively, the input and output shafts. The gear wheels 32, 33 and 34 are coaxial and interlocked to the shaft S. The gear wheels 35, 43, 45 and 46 are coaxial to the intermediate shaft, reference numeral 47. The gear wheel 32 meshes the gear wheel 46, and the gear wheel 46 and the gear wheel 34 mesh with the gear wheel 43. The pair of gear wheels 34 and 43 correspond to the first gear and the pair of gear wheels 32 and 46 correspond to the second gear. The reference numeral 44 indicates a sleeve with synchronization elements to select one of the two above gears. The gear wheel 45 meshes with an intermediate gear wheel (not illustrated) which, when also meshes with the gear wheel 33, permits to obtain the reverse gear. The gear wheels 35 and 36, meshing between them, form the output gear wheel train. Over the gear wheel 36 and in a coaxial and interlocked position with it, there is the satellites carrier 37 of a differential group, which outputs to operate the driving wheels as per the references numeral 39 and 42. The references numeral 38 and 41 correspond to the planetary gears and the reference numeral 40 to the satellites of this differential group.

Figure 24 shows the kinematic diagram of the stepless transmission shown in Figure 23, fitted on a motor vehicle equipped with an engine with a maximum operating range of 5.500 r.p.m. Additional details on this example will be explained later.

Figure 25 is a schematic representation that corresponds to a stepless transmission integrated by the "Module of continuity and power deviation" of Figure 3, with a conventional gear box achievement 48, four transmission ratios forward (i.e. four gears) and one reverse gear. The references numeral 49 and 58 indicate the gear wheels of the so-called constant meshing train. The references numeral 52 and 55 indicate the gear wheel pair of the first gear ; the references numeral 51 and 56 indicate the gear wheel pair of the second gear and the references numeral 50 and 57 indicate the gear wheel pair of the third gear. The fourth gear, i. e., the direct gear, is obtained by shifting the sleeve 60 to the left, so that, the shaft where is fitted the gear wheel 49, that is the input shaft to the conventional gearbox, and the output shaft S are interlocked. The reference numeral 60 indicates, therefore, the selector sleeve for the third and for the fourth gears, and the sleeve 61 allows the selection of the first and the second gears. Finally, the gear wheels pair 53 and 54 correspond to the reverse gear (the intermediate gear wheel has not been represented in order to clarify the drawing). The reference numeral 59 indicates the intermediate shaft where are fitted coaxially and interlocked the gear wheels already described as references numeral 54, 55, 56, 57 and 58.

Figure 26, as well as Figures 23 and 25, is a schematic representation that shows another example of a

stepless transmission achievement. In this case, the stepless transmission is formed by the engagement of the gearbox 62 and the " Module of continuity and power deviation " of Figure 1 (in a reversed position respect to the Figure 1) where, only for the sake of clarity, the trunco-conical pulleys 1 and 2 and the epyciclic gears 11 and 16 of Figure 1 have been represented. With respect to the assembly 62, letters E and S indicate, as always, the input and output shafts respectively. The gear wheels 63 to 68 are coaxial to the shafts E and S that are alined. On the other hand, the gear wheels 69 to 74 are coaxial to the shafts 75 and 76, wherein the second one is of tubular configuration and both are coaxial. The gear wheel 74 is coaxial and interlocked to the shaft 75 and the gear wheels 69 to 73 are coaxial and interlocked to the tubular shaft 76, which, in turn, is coaxial and interlocked to the satellites carrier of the planetary gear 16. The references numeral 77 and 78 indicate the gear selector sleeves. The gear wheels pair 63 and 74 form the known meshing constant gear train; the remaining gear wheels pairs correspond to the following gears; the references numeral 70 and 67 for the first gear, the references numeral 66 and 71 for the second gear, the references numeral 65 and 72 for the third gear, the references numeral 64 and 73 for the fourth gear and the references numeral 68 and 69 for the reverse gear (the intermediate gear wheel, reverse gear characteristic, has not been represented).

Although it could be deduced from the above, it is convenient to outline that the essence of the present invention lies in the fact that the mechanical connection or connections among the different pieces elements and/or mechanical assemblies well known, as the pulleys, epyciclic groups, clutches, brakes, synchronization sleeves, gear selectors, etc. permit to dispose of one assembly, called " Module of continuity and power deviation ", henceforward will be called, to abbreviate, Module of continuity that constitutes the fundamental and basic part of the present invention. Its engagement to gearboxes achievements, some of them are conventional, results in that the assembly in this way built up operates as a stepless transmission with the correct application field to be applied to the automotive industry or to the industry in general.

It is convenient then, to describe first the Module of continuity operation. The appended schematic drawings in Figures 1 to 6 are achievement examples in the same way as the assorted versions that can be obtained grouping together an assembly of Figures 10 and 11 with another one of the Figures 12, 13 and 14 and finally with another assembly taken among Figures 15 to 20. That is to say a total of $2 \times 3 \times 8 = 48$ variants, besides those that can be derived from the Figures 7, 8 and 9, on which the Modules of continuity are based and already mentioned on Figures 1 to 6. Figures 3, 4 and 6 show adequated structures for longitudinal engine arrangements (that is the case in the commercial vehicles and in some passenger cars), whereas, Figures 1, 2 and 5 are preferred arrangements for transverse engines assemblies (that is the most frequent case in passengers cars), although they can also be used for longitudinal engines arrangements, as the example shown in Figures 25 and 26 (the first one corresponds to a vertical section scheme and the second one to a horizontal section scheme). All these Modules of continuity variations have a common basic feature that could be defined as a universal mechanical law that relates the maximum power D, that is directed through the trunco-conical pulleys (and that is a fraction of the power that arrives to the Module of continuity coming from the engine) with the operating field of the above mentioned Module of continuity, C, (that is the quotient between their higher transmission ratios, e.i. maximum and minimum values). This functional dependence is expressed by the following formula:

$$D = \frac{C - 1}{C + 1} \times 100$$

In which D is quantifed by the engine power percentage.

This formula has been deducted from the appropiated mathematical, kinematics and dynamics developments, submitted to that D is limited to a same maximum value for the rotations, in both rotative senses, of some of the elements of one of the epicyclic gears that are integrated in the Module of continuity; it has been assumed the hypothesis of 100 % on the mechanical efficiency of power transmission through the gear wheels and the trunco-conical pulleys. With reference to the so many repeated variety of the Modules of continuity, it must be understood that the Modules of continuity represented in Figures 1 to 6 (with clutch and brake or with synchronization sleeves or with sleeves and brake, as they are shown in Figures 7, 8 and 9) correspond to the most suitable functional achievements, taken into account the studies developed. Figure 21 is the graphic translation of the above formula that relates D and C.

For the description of the Module of continuity operation and in order to make easy its understanding, Figure 22 graphics must be considered (they have been made with practical values that define the planetary gears and the admissible diameters and reductions of the pulleys) and also the Figure 1.

The variable K, represented in the common X-axle on the two graphics of Figure 22, indicates the quotient between the arcs radii embraced by the "belt" 4 in the trunco-pulleys 1 and 2 (see Figure 1). Provided that, when the brake 9 operates, the rotation direction of the gear wheel 15 is reversed with respect to the shaft E and so the sun rotation direction of the epicyclic gear 16, it has been agreed to assign negative values to K, to be able to differentiate it when the clutch 8 works instead of the brake 9. In this case, the rotation direction of the gear wheel 15 and the sun 16, is the same as the rotation direction of the input shaft E, then the positive sign corresponds to K in accordance with the above agreement. The Module of continuity works, in such a

form that for each K value, the graphics of Figure 22 produce the transmission ratio values T (that is the quotient between the E and S rotation speeds) and the power values D (percentage of the power operating on the shaft E, i.e., the engine power to which is connected) that it is transmitted through the trunco-conical pulleys. When the brake 9 works, the maximum values of T and D are obtained when the radius of the "belt" 4 in the trunco-pulley 1 is the maximum (and minimum in the trunco-conical pulleys 2) and that in this example, the corresponding K value is, considering the above signs agreement, -0,925 and then T = 2,008 and D = 32,5 (these values are not indicated in Figure 22 for clearness)). When the K value is the minimum, in this example K = - 0,16, the T and D values are T = 1,582 and D = 4,43, which indicates that, if the brake 9 continues to be applied, the radius continuous variation of the "belt" 4 in the trunco-conical pulley 1, from its maximum value to its minimum value (and the corresponding radius increase of the "belt" 4 in the trunco-conical pulley 2, from its minimum value to its maximum value) results in that the T values are reduced from T = 2,008 to T = 1,582. When the brake 9 stops to operate, in the representative points corresponding to K = - 0,16 and the clutch 8 starts to operate, the operation is such as the new K value to be considered (mantaining the minimum and maximum radii, respectively, for the belt 4 in the trunco-conical pulleys 1 and 2) is K = 0,16, for which, the value T = 1,3998 and D = 7,61, a continuity step is produced in the module transmission ratio value, which value is 1,582/1,3998 = 1,13, i. e., a 13 %, value that is perfectly admissible. With the mentioned clutch 8 applied, if the values of the radii of the "belt" 4 of the trunco-conical pulley 1 are increased from its minimum value to its maximum value (i.e. increasing the K values from 0,16 to 0,925), the T values are decreased from 1,3998 to 1,026, instead the power values of D are increased from the initial value D = 7,61 to the maximum one D = 32,3 (this last value is practically equal to the maximum value that corresponded to K = - 0,925 for D = 32,5). In short, the module tramsmission ratio has taken all the values, in a continuous manner, from the maximum value T = 2,008 to the minimum one T = 1,026 which is equivalent to an actuation field C of value C = 2,008/1,026 = 1,9571, whereas the power D has been mantained in values lower than 32,5 % in all the interval of K variation, except in the point K = -0,925 in wich D = 32,5 %. It is understable that the association of this Module of continuity and a conventional gearbox achievement which transmission relations have numerical values in geometrical progression of ratio 1,9571, i.e., the corresponding value to the C operation field of the Module of continuity, results in the availability of a stepless transmission without steps or lack of continuity. The function played by the mentioned brake 9 and clutch 8, can be replaced with the appropiated synchronization sleeves 24 shown in Figures 7 and 8. In Figure 7, the shift of the synchronization sleeve 24 to the right side produces the locking of the satellites carrier 26 (this function is equivalent to the already mentioned function of the brake 9) and in its shift to the left side produces the interlocking of the satellites carrier 26 and the sun 10, which is equivalent to the clutch 8 operation. The speeds to achieve the connections of the above sleeve are admissible for its adequated operation. In Figure 8 the clutch function is achieved by shifting the synchronization sleeve 24 to the left side and it is not possible the shift to the right side from the position of the training gear wheel 23, since here the brake 9 is mantained. Figure 23 with the Module of continuity corresponding to the example used for the above explanation and with the gearbox, as indicated, with two transmissions relations forward, corresponds to a gearbox scheme for applications on passenger car equipped with transverse engine and its kinematic behaviour is as shown in Figure 24. In order to value its operation quality, it will be sufficient to consider that in the first gear there are two continuity areas of their tansmission relations, the first one comprised between the values 3,405 and 2,683 and the second one between the values 2,374 and 1,740; the step between them is 2,683/2,374 = 1,13, the same value, as it is obvious, that the discontinuity of the module operation. In the second gear there are also another two areas, one of them comprised between the values 1,549 and 1,221 and the other one between the values 1,08 and 0,791; (the step between them is 1,221/1,08 = 1,13. In short, the actuation field of this gearbox is 3,405/0,791 = 4,305 and the steps or discontinuity solutions among its four areas wiil be: <u>1,13</u> : 1,740/1,549 = <u>1,12</u> and 1,13. If the maximum power of the motor vehicle engine is, for example, 110 C.V., the maximum power transmitted through the trunco-conical pulleys of the Module of continuity is (32,5/100) x 110 = <u>35,75 C.V.</u> and the minimum one is (4,43/100) x 110 = <u>4,873 C.V.</u> For the above and assuming the hypothesis of equality of utilization times, it can be said that the deviated average power through the trunco-conical pulleys has a value of (33,75 + 4,873) / 2 = 20,2965 C.V. ≈ <u>20.3 C.V.</u> In the case that an arrangement stepless transmission would ben used, based on the trunco-conical pulleys without conection with epicyclic gears, i.e., just as the arrangements mentioned at the beginning, passenger car Fiat Uno Selecta or similar types, the transmitted power by the above trunco-conical pulleys would be <u>110 C.V.</u>

If two pairs of trunco-conical pulleys were being used, instead of only one pair, just as the modules used in Figures 5 and 6 and the K values for each pair of trunco-conical pulleys were the same, anyway its lower value K = 0,16, the discontinuity among adjacent areas corresponding to the mechanical relations 1st and the 2nd, would have the value 1,021 instead of the above mentioned value 1,13. That is to say, almost a complete continuity that, possibly and practically, is not worthwhile due to the cost and pomplexity which that involves.

Just before finishing this exposition, it is con venient to consider that the planetary gears 11 and 16 can al-

ways be defined in such a way that the value of the power deviation, D, can be the same practical maximum value for a same K value, corresponding to one or another sun rotation direction of the gear 16, i.e., the satellites carrier of 11 free or immobilized, now for a brake now for a sliding synchronism sleeve. Naturally, the K maximum values can also be determined, one of them positive and the other one negative in accordance with the signs agreement already explained, in order to fulfil the values equality of the maximum power transmitted by the trunco-conical pulleys.

It is also convenient to outline that for motor vehicles equipped with high powered engines, it is possible to achieve a stepless transmission such as, through the trunco-conical pulleys of its Module of continuity could be deviated a maximum power value coming from the engine that not exceed the value of 58 C.V. mentioned at the beginning of this specifications. In fact, if for example the engine maximum power of the motor vehicle is 300 C.V. and a value of 5,2 is necessary for the field of application of the stepless transmission (the present passenger cars have conventional gearboxes of five gears and such as their application fields, are placed, approximately, between 4 and 5,5), it is possible to define a stepless transmission, based in this invention patent such as the achievement that incorporates a conventional gearbox has only 3 transmission relations (instead of the present 5) and that through the trunco-conical pulleys of its Module of continuity can transmit a maximum power of <u>58 C.V.</u> The kinematic diagram (passenger car speed and engine r.p.m.) would have three areas, each of them divided in two areas due to the Module operation discontinuity as it was already explained and such as, for example, the discontinuities between the extreme area, that corresponds to each mechanical transmission relation, and the adjacent area, will be 1,30 between the 1st and the 2nd gear and 1,24 between the 2nd and the 3rd gear. If a four transmission relations would be used instead of one of three conventional transmission relations arrangement and at priori were selected, a discontinuity steps, of the same values, between the adjacent areas of the kinematic diagram, 1,25, the maximum power deviated and transmitted by the trunco-conical pulleys would be <u>36,6 C.V.</u> These values of 58 C.V and 36,6 C.V, for the above examples, would be converted in 58/3 = 19,33 C.V. and 36,6/3 = 12,2 C.V. respectively in the hypothesis that the maximum power of the engine of the passenger car in question was 100 C.V. All the above justifies all that has been exposed at the beginning of this text; for medium and lower power engines, stepless transmissions can be built-up using the trunco-conical pulleys and conventional "belts" traction operating and for higher power engines using the trunco-conical pulleys and the mentioned metallic "belts", unless it was choosen the use of conventional achievements of the conventional gearbox, integrated on the stepless transmission, with a higher number of mechanical transmission relations in order to reduce, until achieving the appropiated value, the value of the maximum power deviated and transmitted by the trunco-conical pulleys. Figure 27 shows an achievement variation of the power transmission mechanism. This mechanism comprises a training gear wheel 80, that is interlocked to the shaft 3 and over which there is a sliding sleeve 81 with synchronization mechanism. Besides, on shaft 3 there are mounted a gear wheel 5 and a gear wheel 82, which can rotate freely with respect to the shaft 3, but they will be interlocked to the shaft 3 when the sleeve 81 shifts to mesh the coaxial training gear wheel that is coaxial and monopiece with the gear wheel 5, or with the training gear wheel that is coaxial and monopiece with the gear wheel 82.

Gear wheel 82 corresponds to the reverse gear of a conventional gearbox when meshed with an intermediate gear wheel 83 that, in turn, meshes with the gear wheel 84 pressed-on the output shaft 85 that is interlocked to the gear wheel 17.

For the rest, the achievement shown in Figure 27 corresponds to the achievement of Figure 1.

Figure 28 shows a second way for the achievement of the power transmission mechanism. In this case, the mechanism comprises a differential group 86, which satellites carrier 87 can be interlocked by shifting the appopriate sleeve 88 with training gearing and synchronism, in which case the rotation direction of its output is overturned.

The above mentioned sleeve 28 has a position in which it interlocks the satellites carrier 87 to the differential mechanism input shaft 89, position that corresponds to the attached transmission relation. The remaining elements and layouts correspond to the achievement showed in Figure 2.

Claims

1. A transmission arrangement including a forward drive and a reverse drive, a V-belt-type continuously variable transmission with at least a pair of pulleys (1, 2) connected by a flexible transmission element (4), a power transmission mechanism (11,5, 17,82, 83, 84, 85, 86) and an output planetary gear train (16) composed of a carrier (13) for a plurality of planet gears (14), a sun gear and a ring gear (12), whereby one pulley (1) of said pulleys is connected to a transmission input shaft (E) receiving power from an engine and the other pulley (2) is drivingly connected to an input shaft of said power transmission mechanism which has an output gear (15) drivingly connected to said sun gear of said output planetary gear train (16), whereby one element (12 or 13) of said ring gear (12) or said carrier (13) of said output planetary gear train (16) is drivingly connected to said transmission input shaft (E) and the other element (13 or 12) is connected to an output shaft (S) of -the transmission arrangement, which shaft

(S) being an input shaft of a stepped conventional gearbox, characterized in that said power transmission mechanism is provided with a forward/reverse drive shift mechanism (8,9,23,24,80,81,88) so that upon its actuation said sun gear of said output planetary gear train (16) rotates either in one or in the opposite direction.

2. Transmission according to claim 1, characterized by the fact that the forward/reverse drive shift mechanism consists of a clutch (8, 24) and a brake (9) that respectively act for the said gears.

3. Transmission according to claim 1, characterized by the fact that the forward/reverse drive shift mechanism consists of a displaceable sleeve (24) which, displaced in one direction engages toothed wheels (23, 22), and in the opposite direction brakes the toothed wheel (23) when acting on a fixed element (25).

4. Transmission according to claim 1, characterized by the fact that the forward/reverse drive shift mechanism consists of a sleeve (81) capable of optionally engaging a wheel (80) that is integral with the shaft of the pulley leading to one gear train (5, 17, 15) or to another gear train (82 83, 84, 17, 15).

Patentansprüche

1. Transmission, einschliesslich einen Vorwärts- und einen Rückwärtsgang, eine stufenlose regelbare Keilriementransmission mit wenigstens einem mittels eines flexibeln Übertragunselement (4) angeschlossenen Scheibenpaar (1, 2), einem Leistungsübertragungswerk (11, 5, 17, 82, 83, 84, 85, 86) und einer Planetenrädergetriebe (16) für Ausgangsleistung, bestehend aus einem Schlitten (13) für verschiedenen Planetenrädergetriebe (14), einem Planetenzahnrad und einem Zahnkranz (12), wobei eine der genannte Scheiben (1), die die Leistung aus einem Motor bekommt, an einer Eingangsübertragungswelle (E) und die andere Scheibe (2) antreibend an einer Eingangswelle von diesem o.g. Leistungsübertragungswerk angeschlossen ist, wobei dieser letzte einem Ausgangsrad (15) hat, das mit dem Planetenzahnrad dieser Planetenrädergetriebe (16) antreibend verbunden ist, wobei ein Element (12 oder 13) dieses Zahnkranz (12) oder dieses Schlitten (13) von o.g. Ausgangsplanetenrädergetriebe mit der Eingangsübertragungswelle (E) antreibend und das andere Element (13 o. 12) an einer Ausgangswelle (S) der Transmission angeschlossen ist, wobei diese Welle (S) ein Eingangswelle einer konventionelle gestufte Getriebe ist, dadurch gekennzeichnet, daß das gennante Leistungsübertragungswerk mit einer Vor-und Rückgangswerkwelle (8, 9, 23, 24, 80, 81, 88) versehen ist, so dass durch ihre Betätigung, dieses Planetenzahnrad der o.g. Ausgangsplanetenrädergetriebe (16) entweder in einer oder in gegengesetzer Richtung dreht.

2. Transmission gemäß Anspruch 1, dadurch gekennzeichnet, daß die Vor-und Rückgangswerkwelle aus einer Kupplung (8, 24) und einer Bremse (9) besteht, die entsprechend auf die genanten Zahnräder wirken.

3. Transmission gemäß Anspruch 1, dadurch gekennzeichnet, daß die Vor-und Rückgangswerkwelle aus einer verstellbare Hülse (24) besteht, die bei Verschiebung in einer Richtung Zahnräder (23, 22) eingrifft, und in der entgegengesetzte Richtung die Zahnrad (23), bei Wirkung auf einem feststehenden Element (25), bremst.

4. Transmission gemäß Anspruch 1, dadurch gekennzeichnet, daß die Vor-und Rückgangswerkwelle aus einer Hülse (81) besteht, die wahlweise ein eine mit der Scheibewelle bildende Einheit Rad (80) eingreifen kann, wobei dieses entweder die eine Zahnrädergetriebe (5, 17, 15) oder die andere Zahnrädergetriebe (82, 83, 84, 17, 15) treibt.

Revendications

1. Dispositif de transmission comprenant une commande avant et une commande arrière, une courroie de transmission en V et à variation continue, avec au moins deux paires de poulies (1, 2) reliées par un élément de transmission souple (4), un mécanisme de transmission de puissance (11,5, 17, 82, 83, 84, 85, 86) et un train d'engrenage planétaire de sortie (16) composé d'un porteur (13) d'un certain nombre d'engrenages planétaires (14), d'une roue solaire et d'un engrenage en anneau (12), auquel l'une desdites poulies (1) est reliée à un axe d'entrée de transmission (E) recevant la puissance d'un moteur et dont l'autre poulie (2) est reliée à l'axe d'entrée dudit mécanisme de transmission de puissance, possédant un engrenage de sortie (15) relié à la roue solaire du train d'engrenage solaire de sortie (16) auquel un élément (12 ou 13) de l'engrenage à anneaux en question ou du porteur du train d'engrenage planétaire de sortie (16) est relié audit axe d'entrée de transmission (E) ; l'autre élément (13 ou 12) est relié à un axe de sortie (S) du dispositif de transmission, l'axe (S) étant un axe d'entrée d'une boîte de vitesses conventionnelle, caractérisé par le fait que le mécanisme de transmission de puissance en question est pourvu d'un mécanisme de changement de direction avant/arrière (8,9,23,24,80,81,88) de sorte qu'il tourne soit

dans une direction, soit dans la direction opposée sur son actionneur dit roue solaire du train d'engrenage planétaire de sortie.

2. Transmission conformément à la revendication 1, caractérisée par le fait que le mécanisme de changement de direction avant/arrière comporte un embrayage (8, 24) et un frein (9) qui agissent respectivement sur lesdits engrenages.

3. Transmission conformément à la revendication 1, caractérisée par le fait que le mécanisme de changement de direction avant/arrière comporte un manchon (24) qui, lorsqu'il se déplace dans une direction s'engage sur des couronnes dentées (23, 22) et lorsqu'il se déplace dans la direction opposée, freine la couronne dentée (23) quand elle agit sur un élément fixe (25).

4. Transmission conformément à la revendication 1, caractérisée par le fait que les mécanisme de changement de direction avant/arrière comporte un manchon (81) capable de s'engager à volonté sur une couronne (80) faisant partie intégrante de l'axe de la poulie et menant à un train d'engrenage (5,17,15) ou à un autre train d'engrenage (82,83,84,17,15).

FIGURE 1

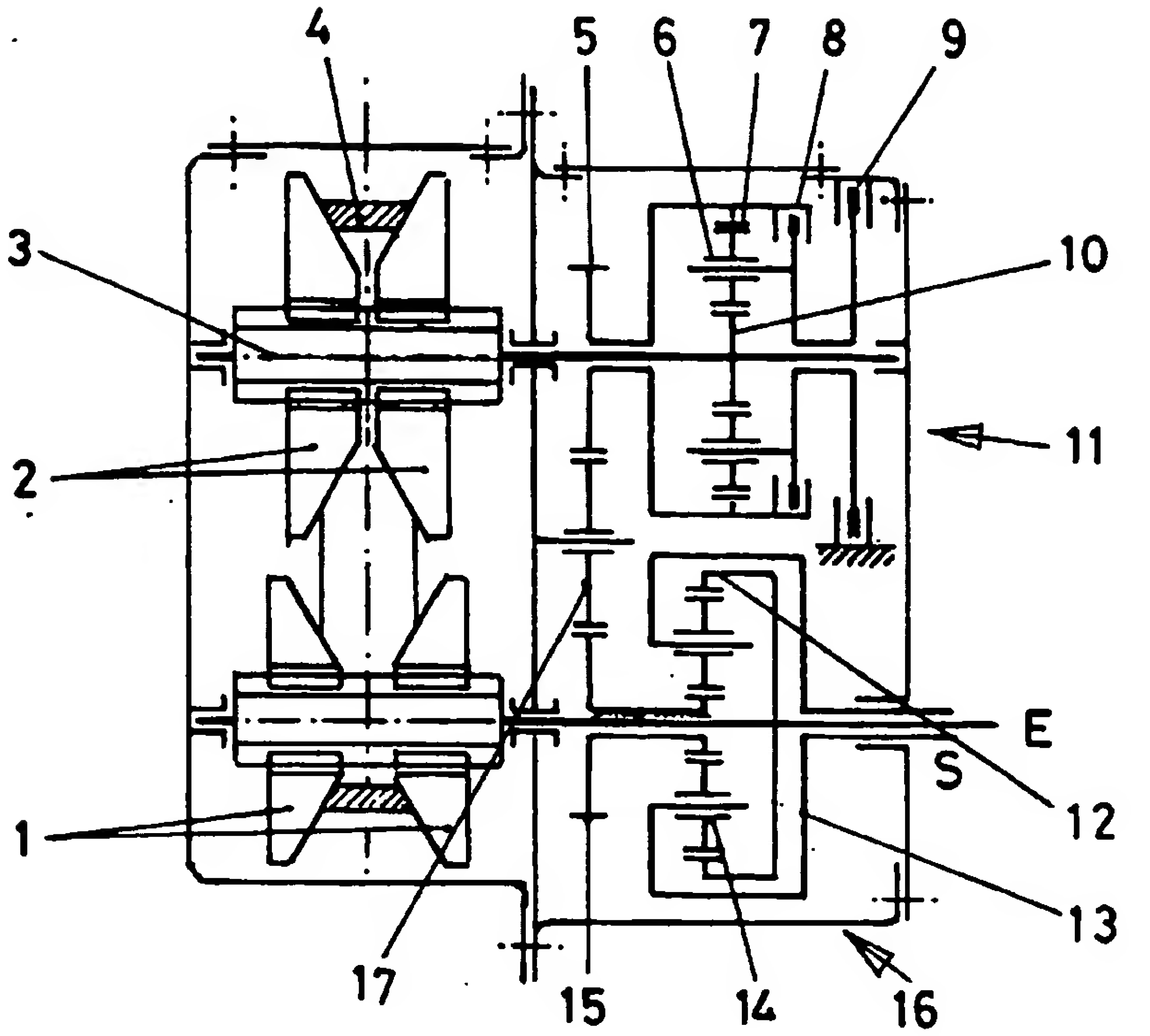


FIGURE 2

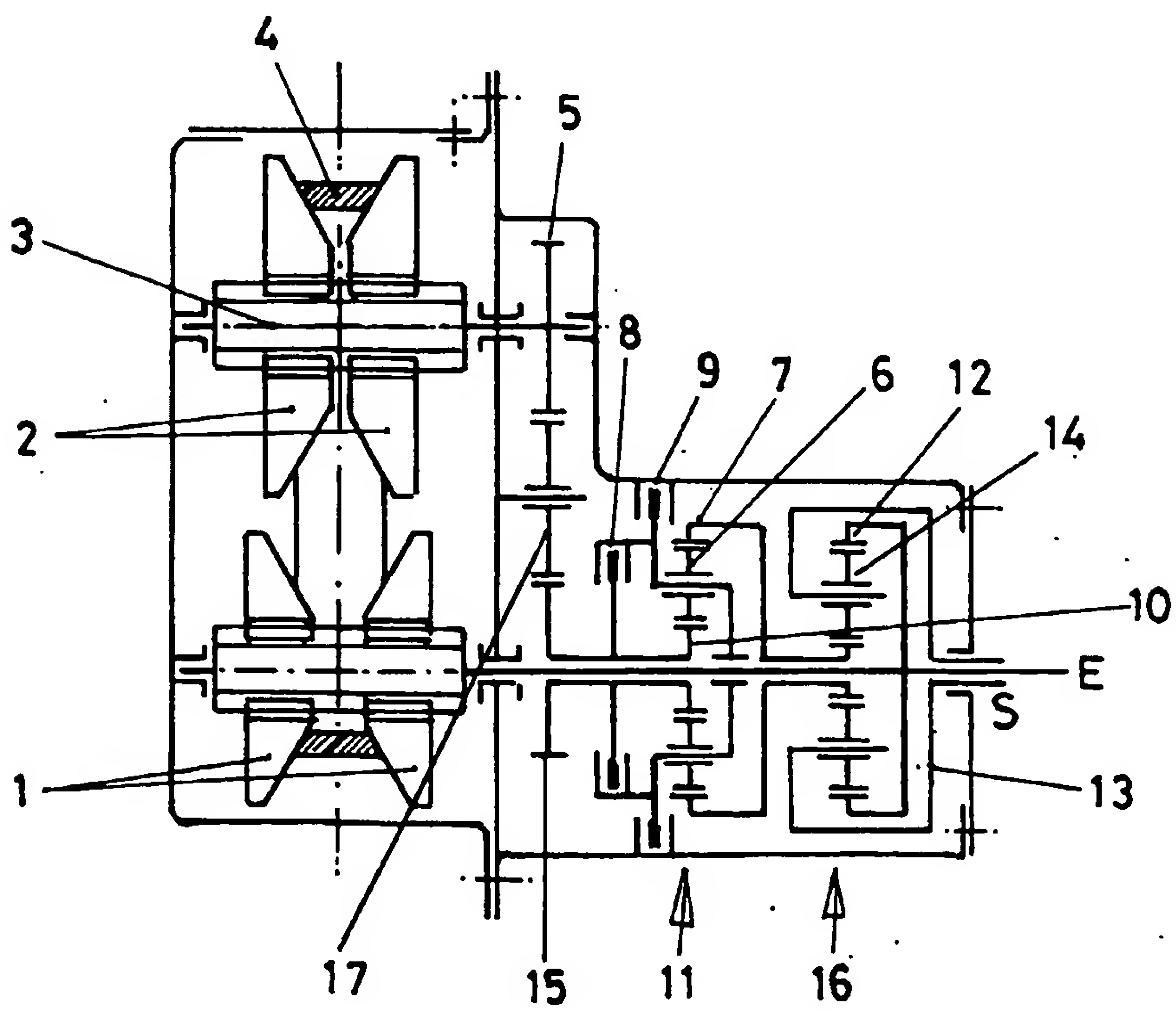


FIGURE 3

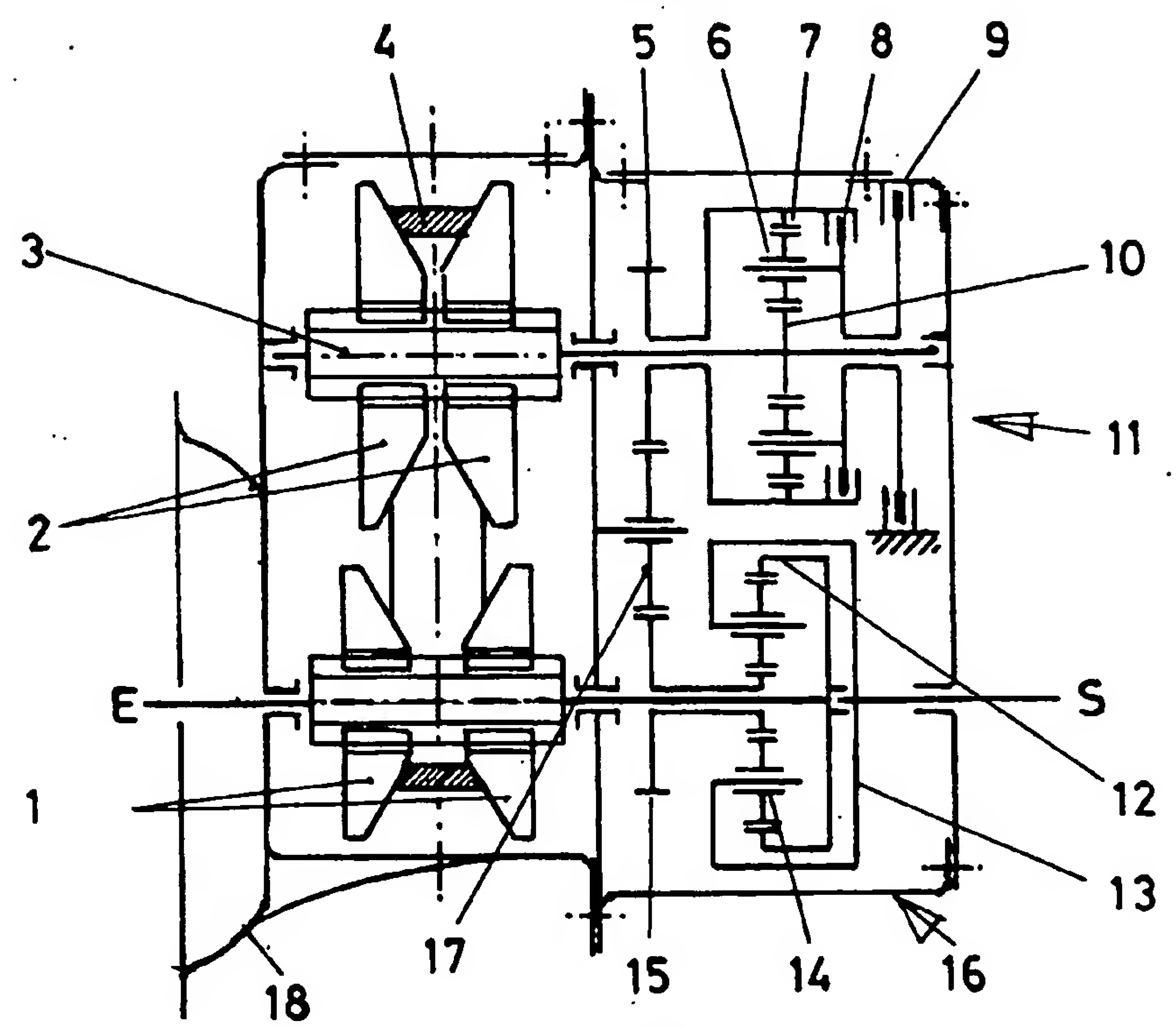


FIGURE 4

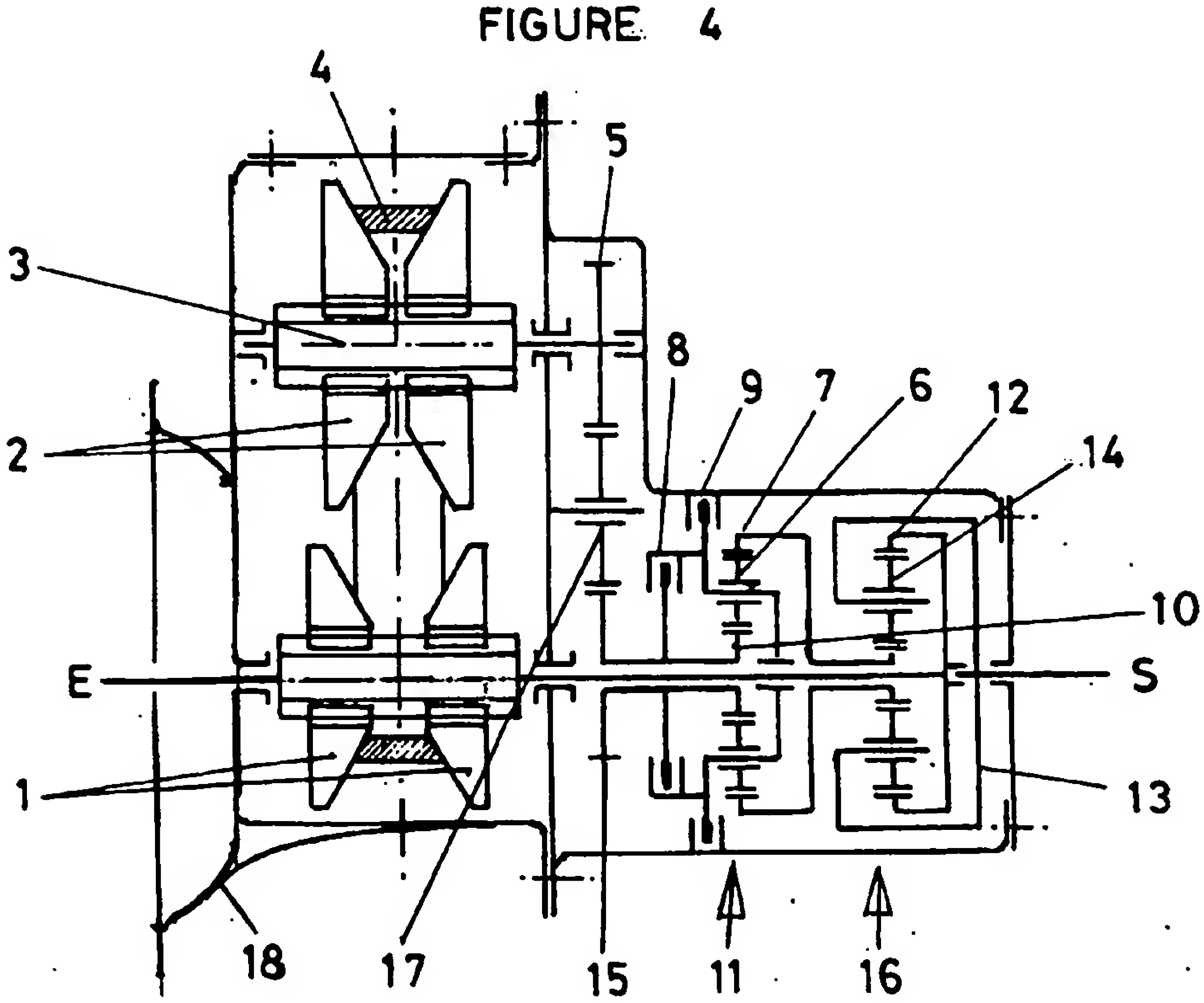
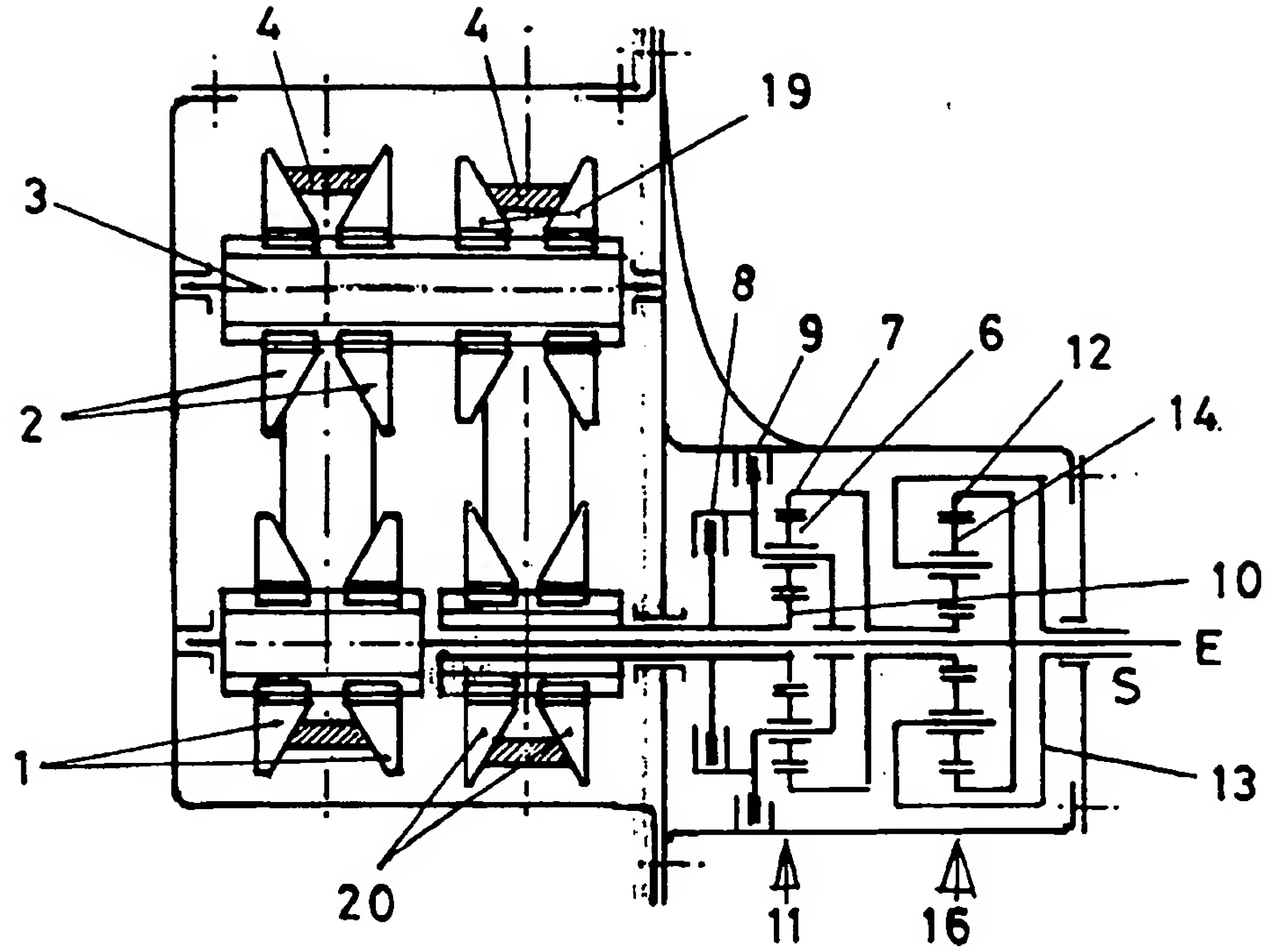


FIGURE 5



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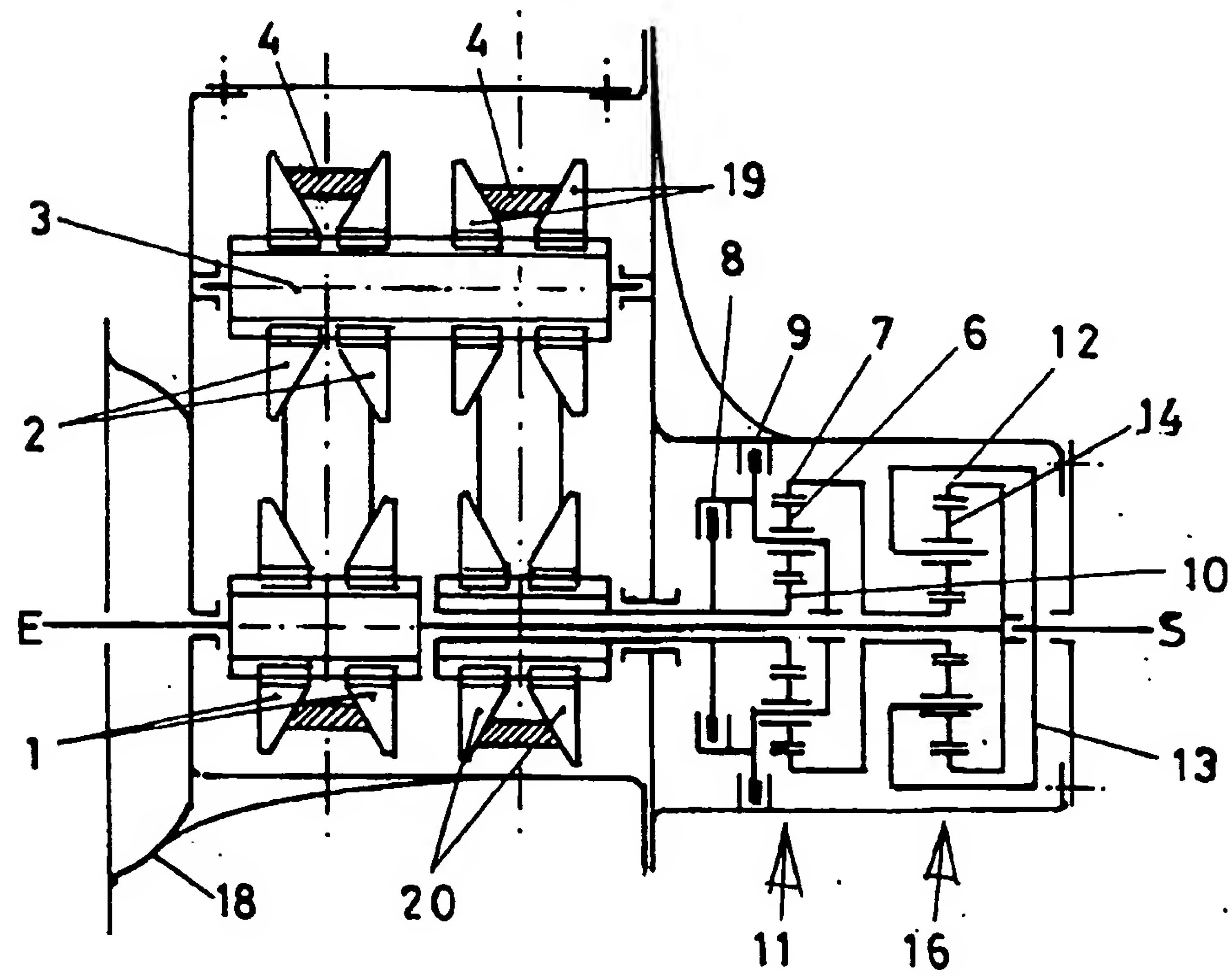

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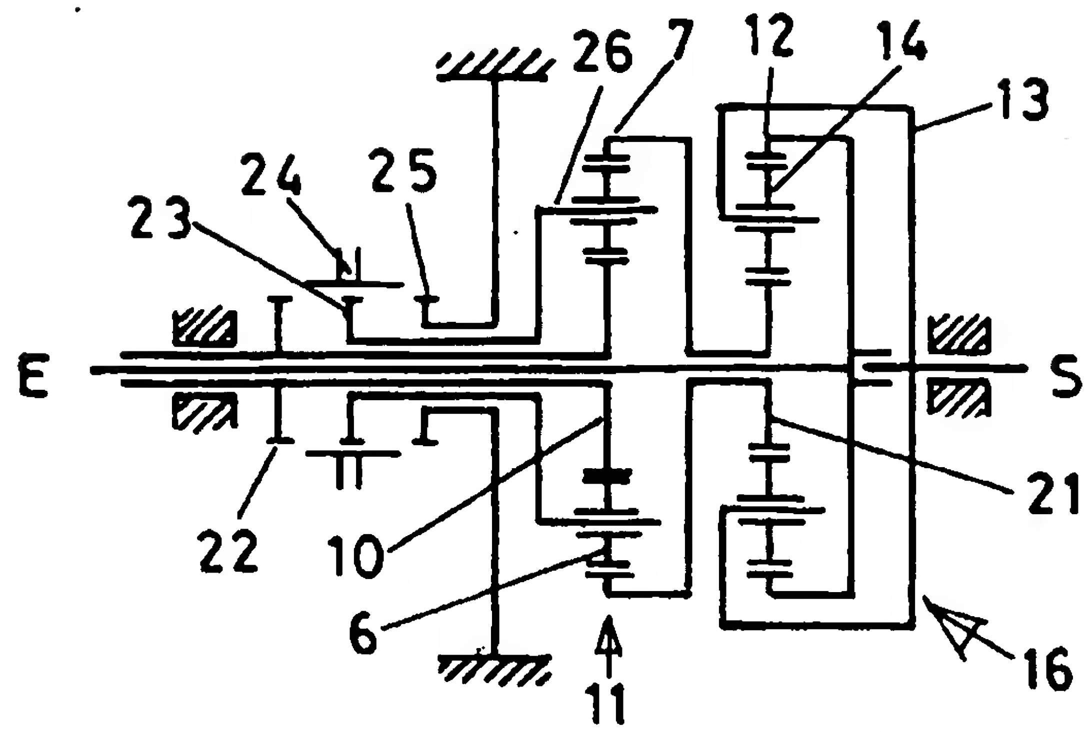

FIGURE 8

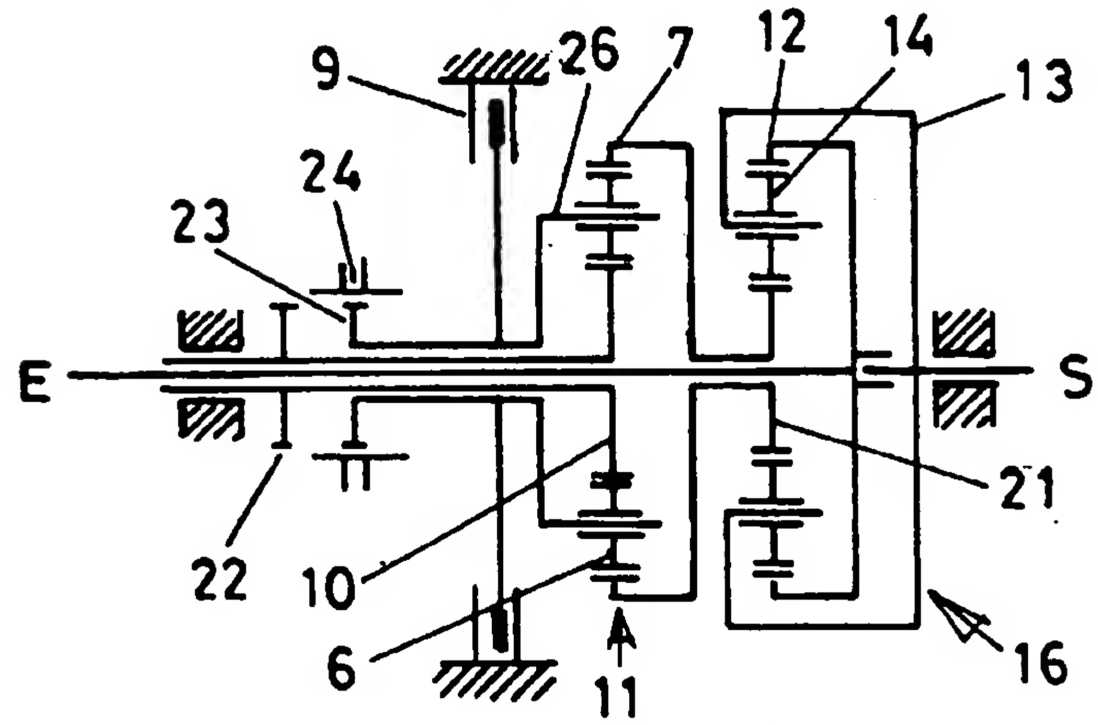

FIGURE 9

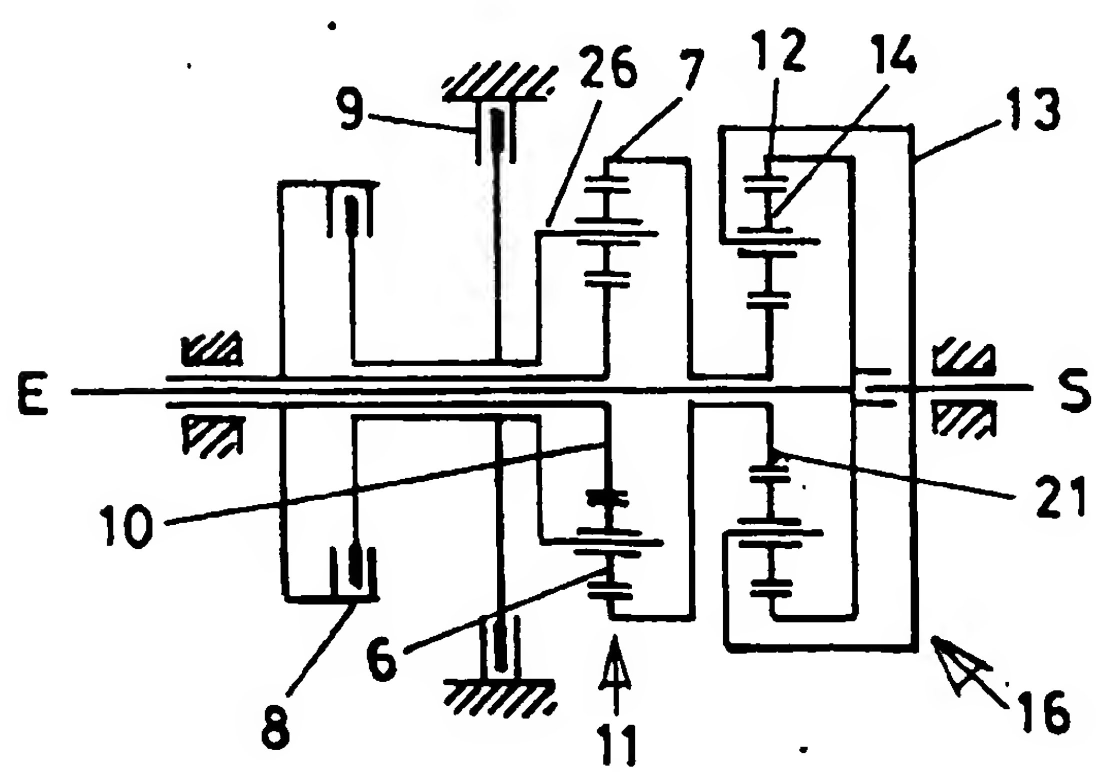

FIGURE 10

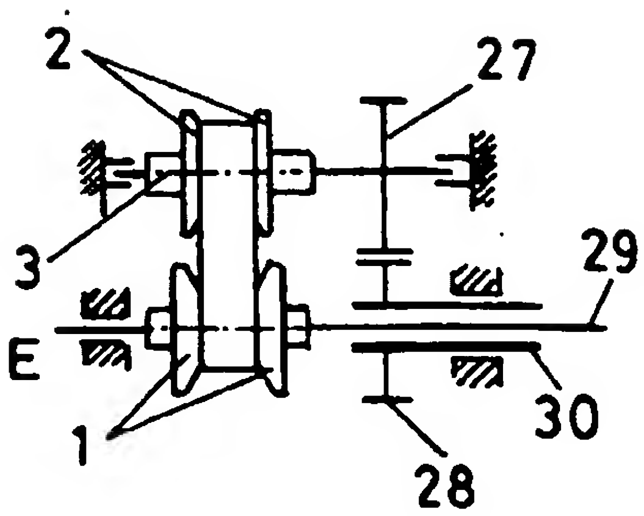

FIGURE 11

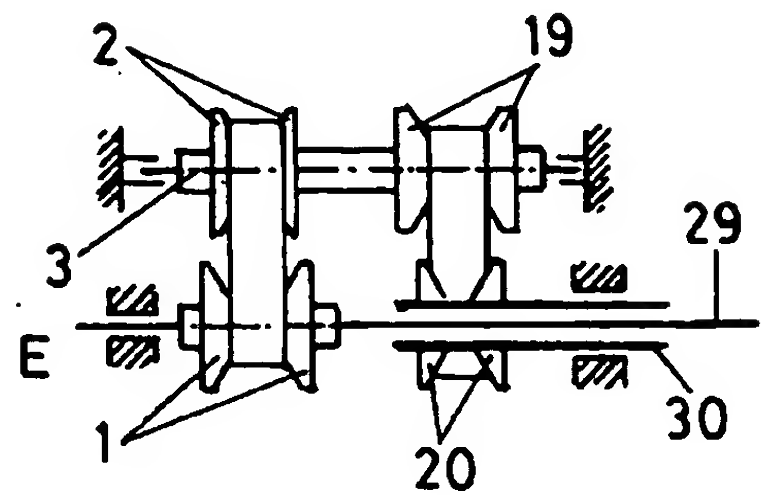

FIGURE 12

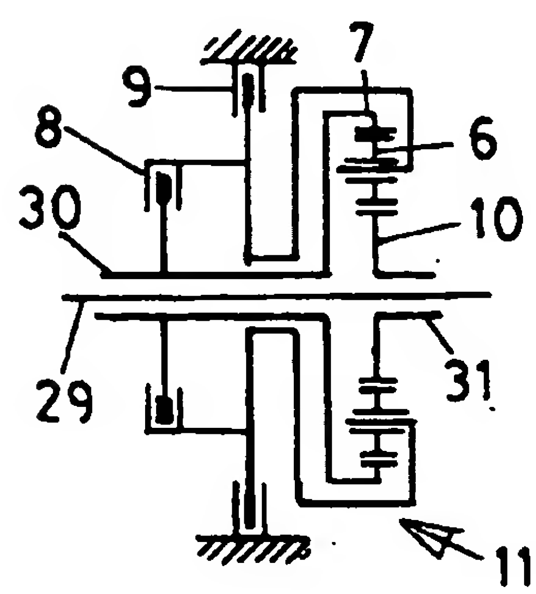

FIGURE 13

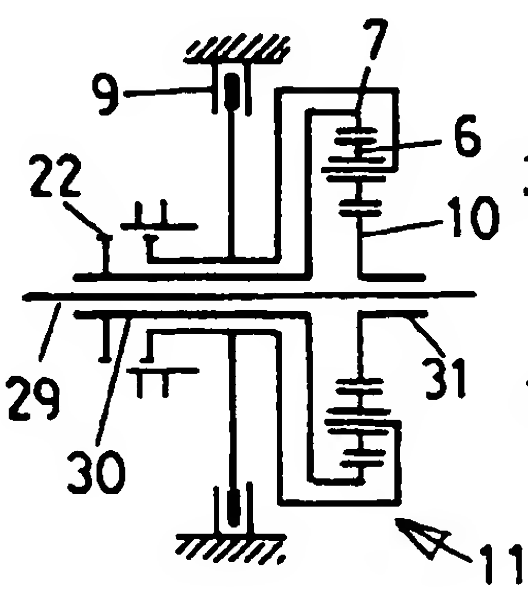

FIGURE 14

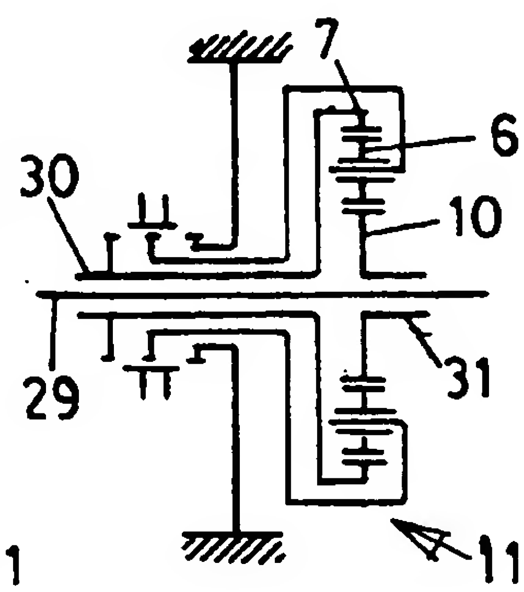

FIGURE 15

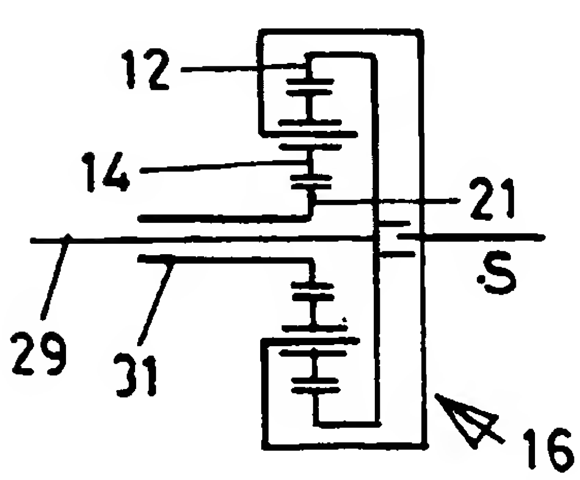

FIGURE 16

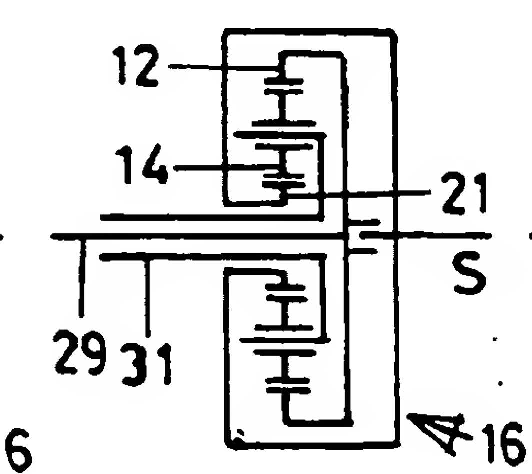

FIGURE 17

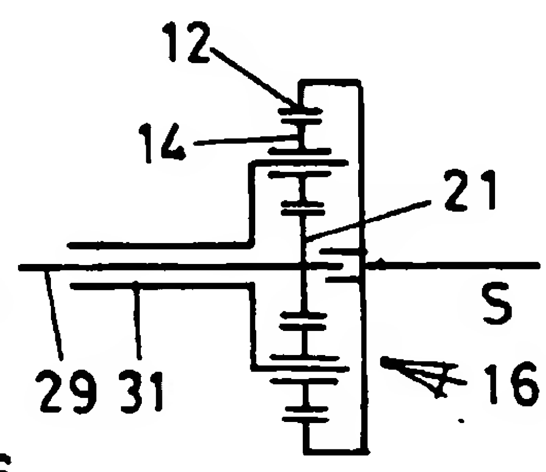

FIGURE 18

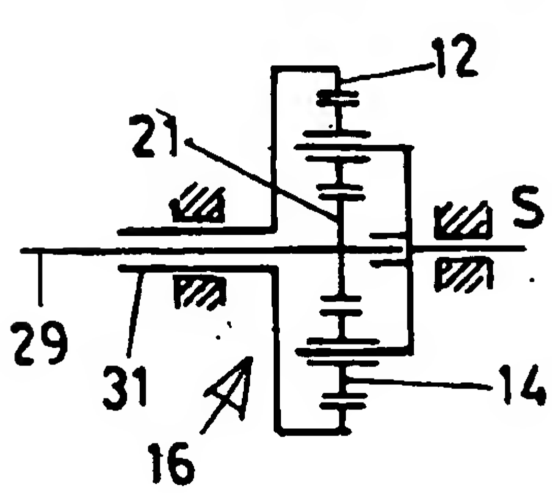

FIGURE 19

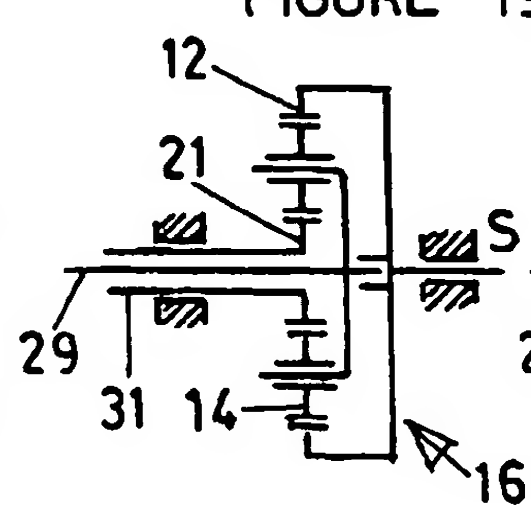

FIGURE 20

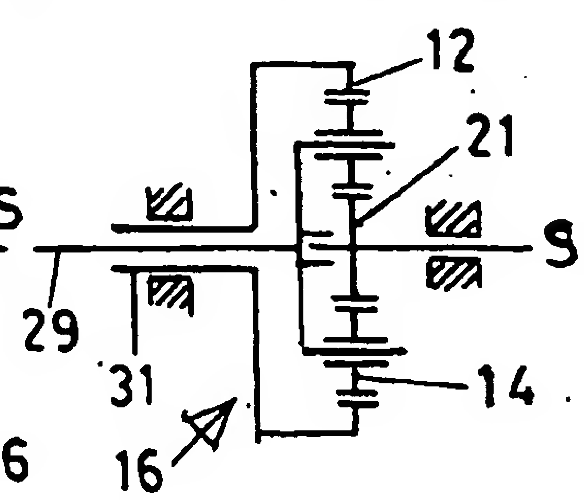

FIGURE 21

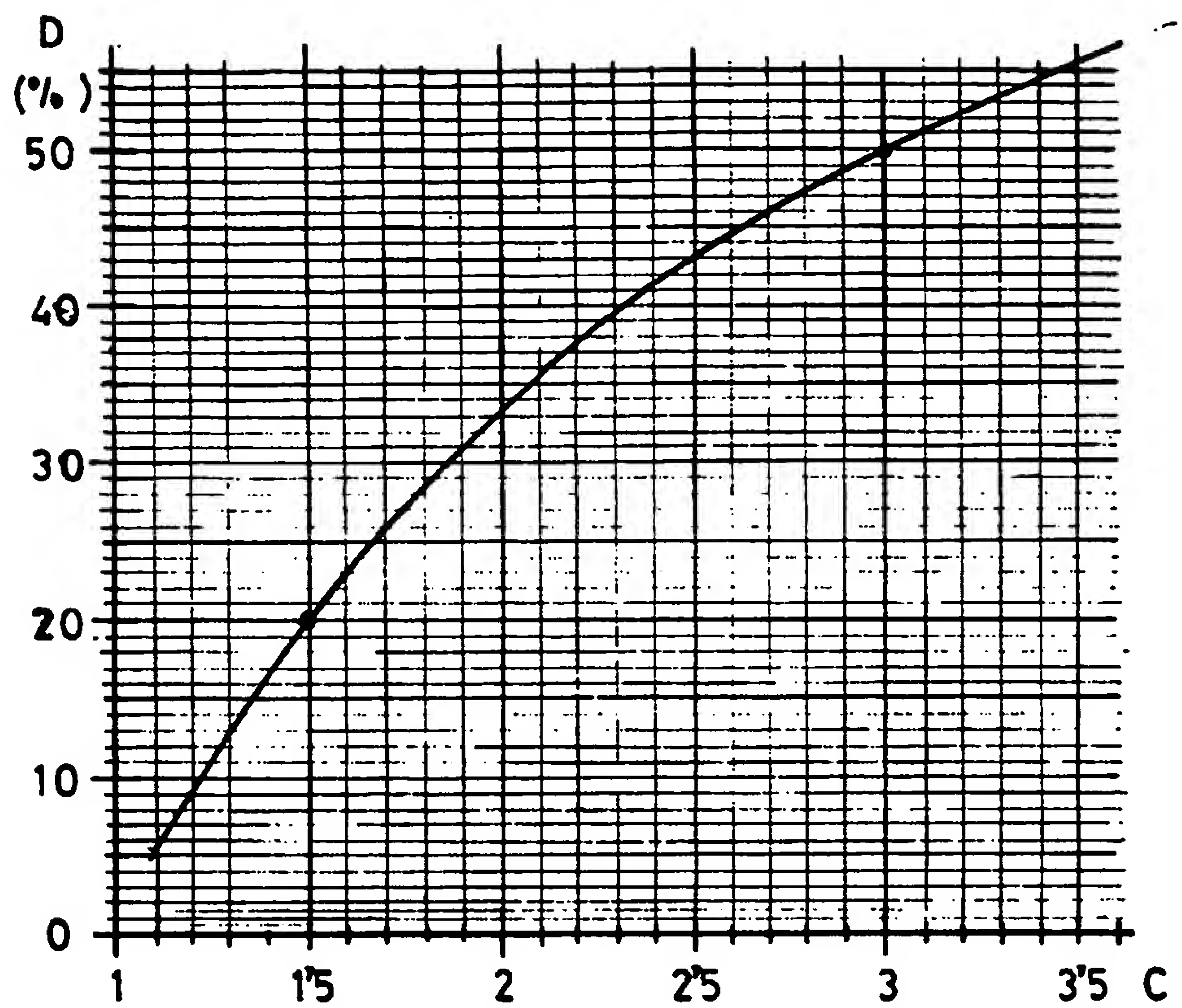

FIGURE 22

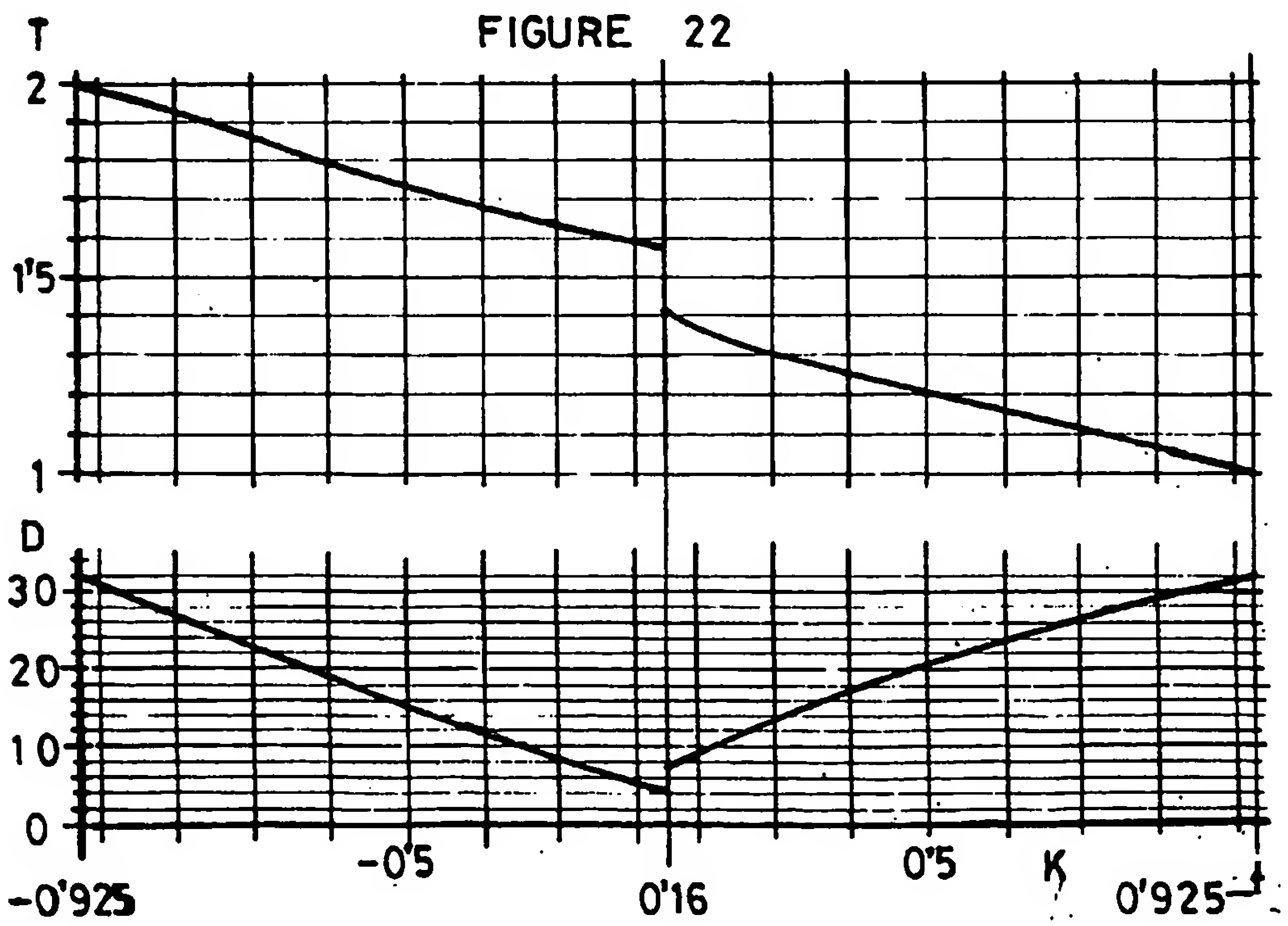

FIGURE 23

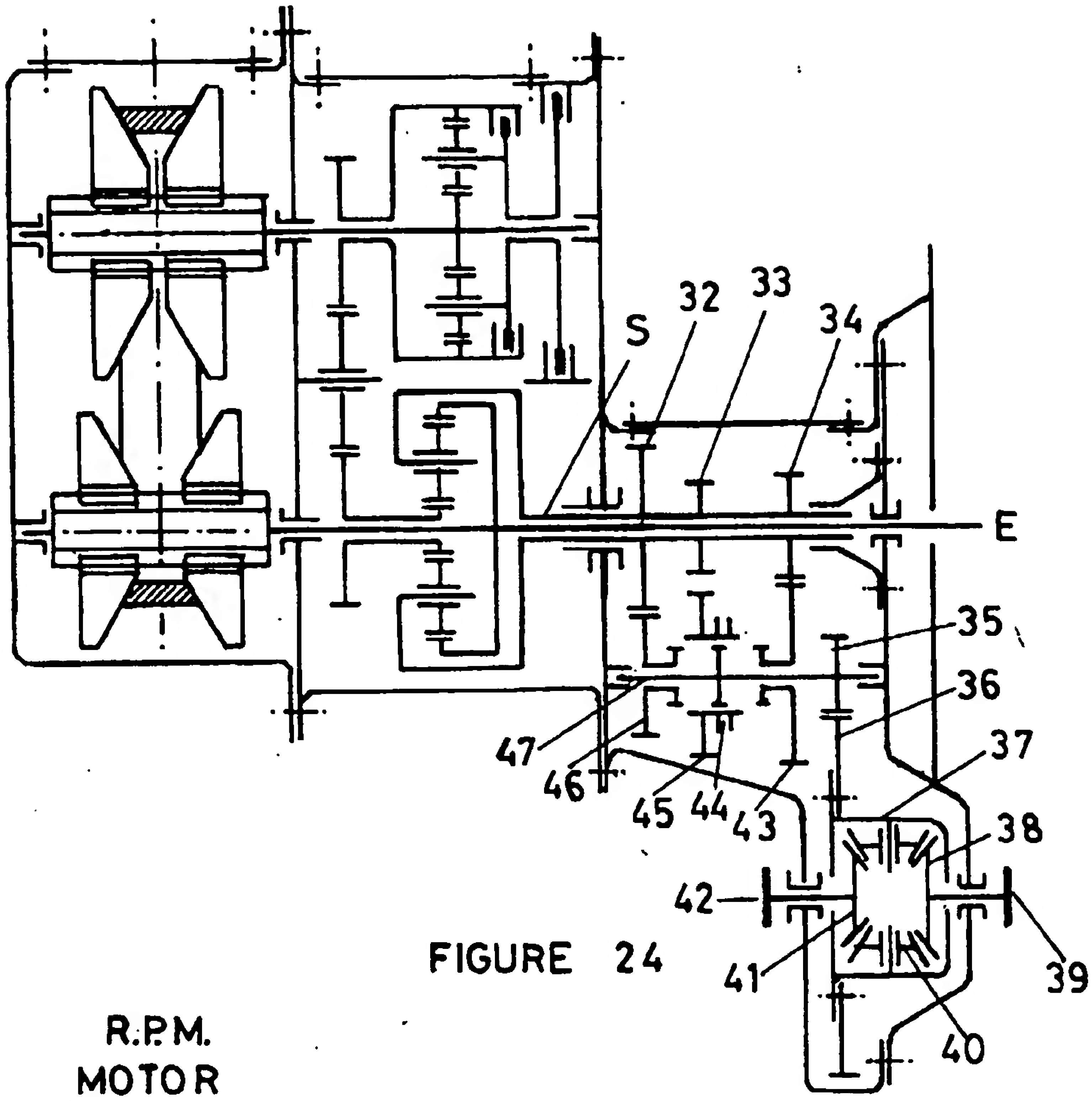

FIGURE 24

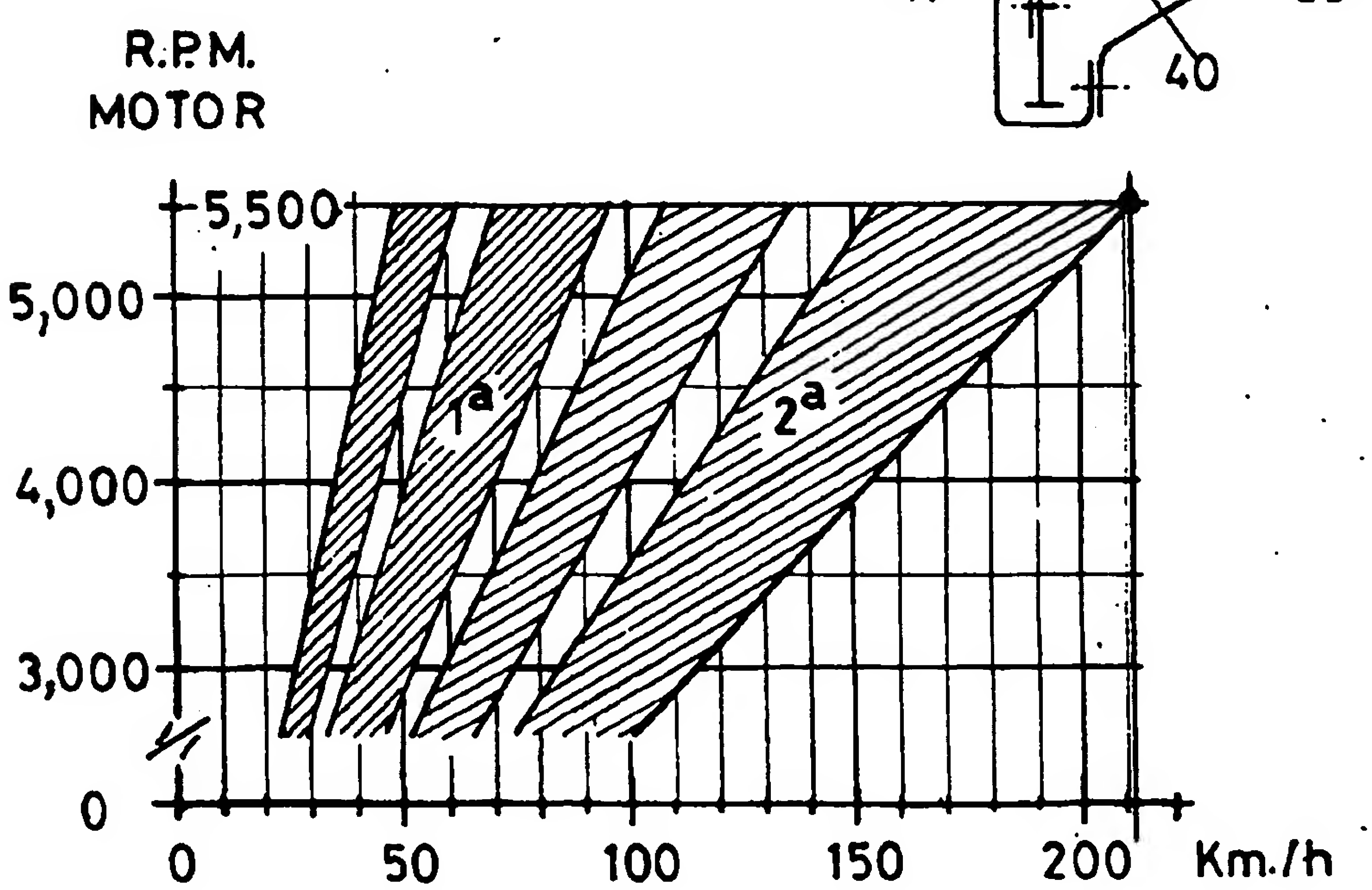

FIGURE 25

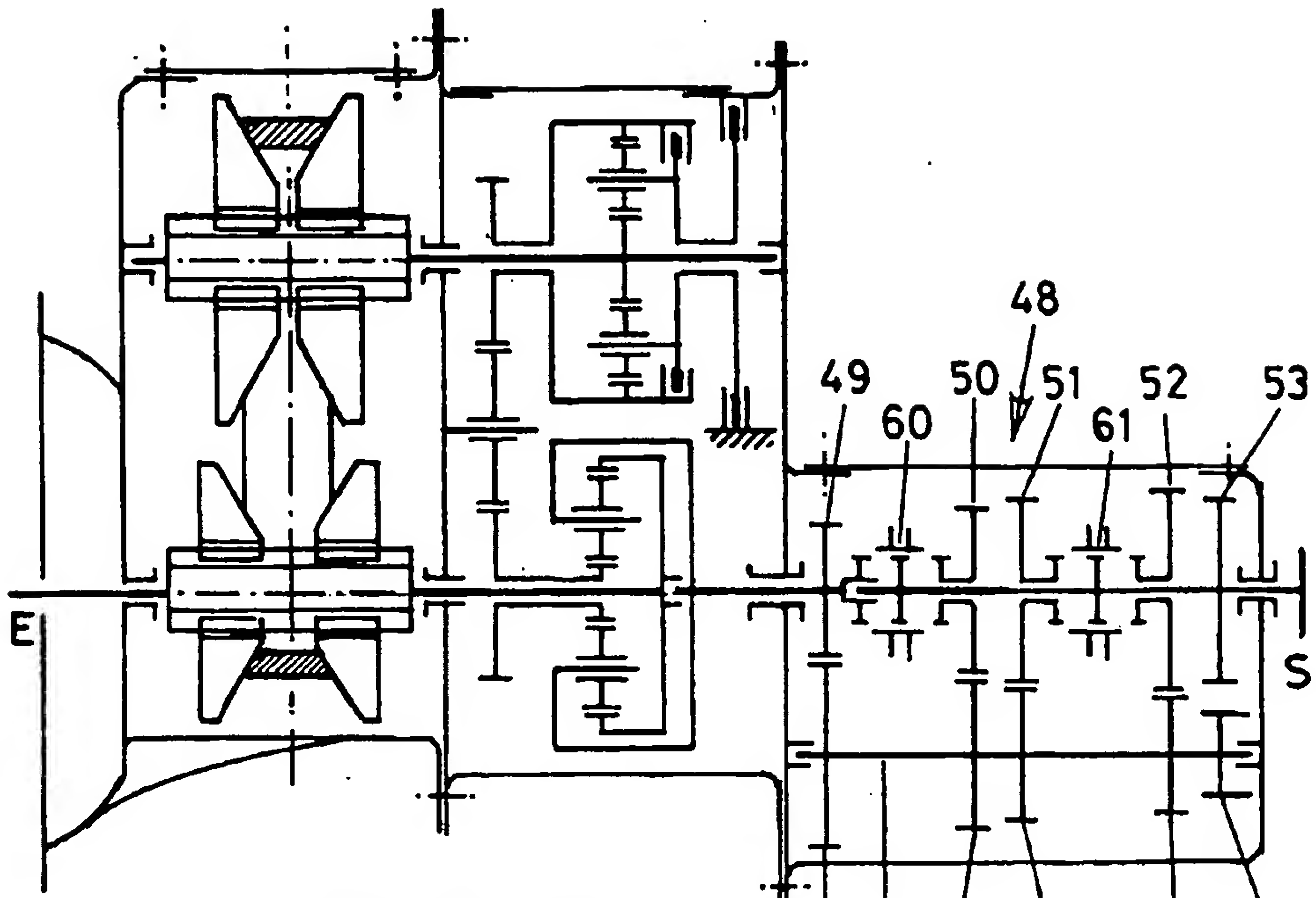

FIGURE 26

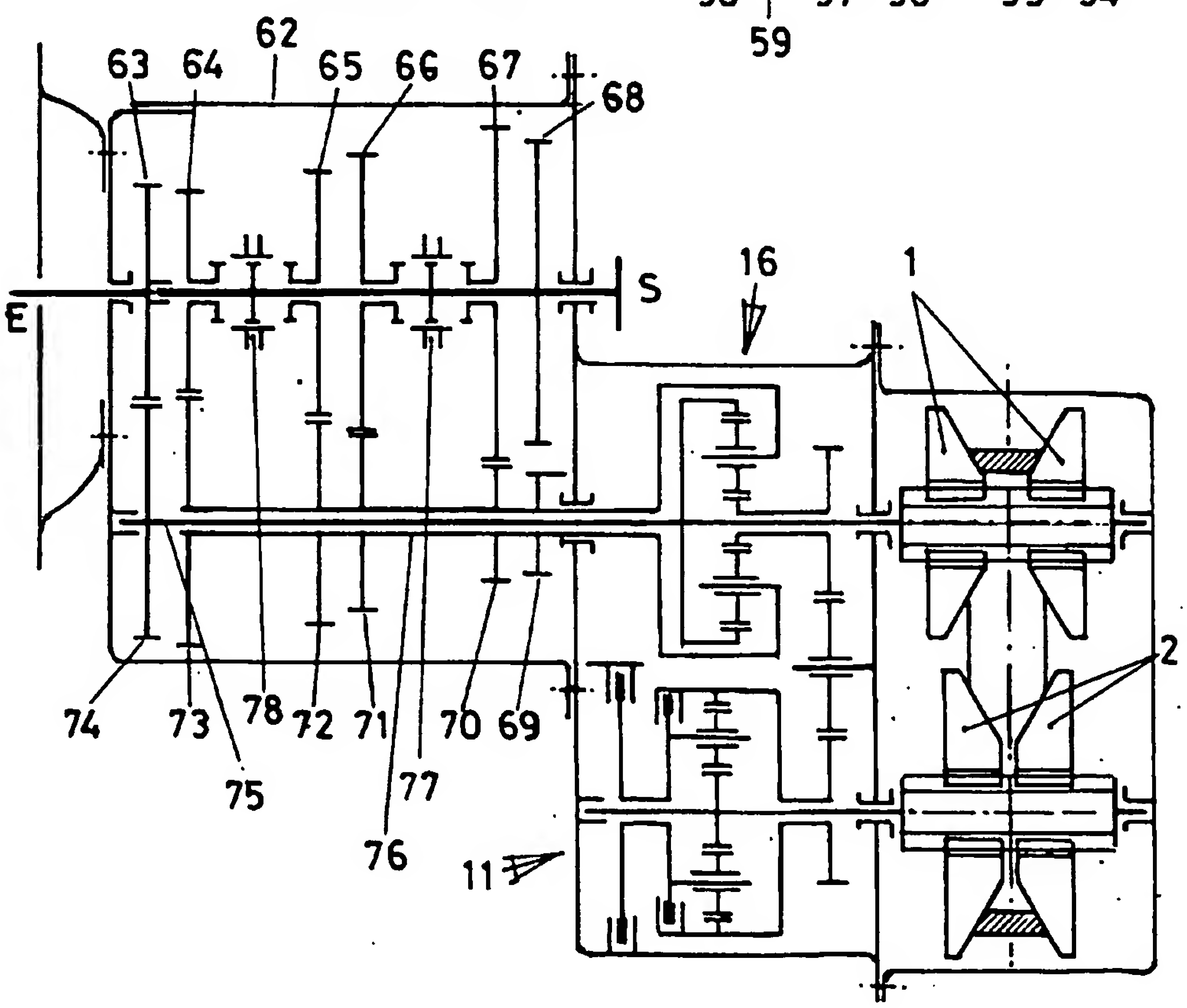

FIGURE 27

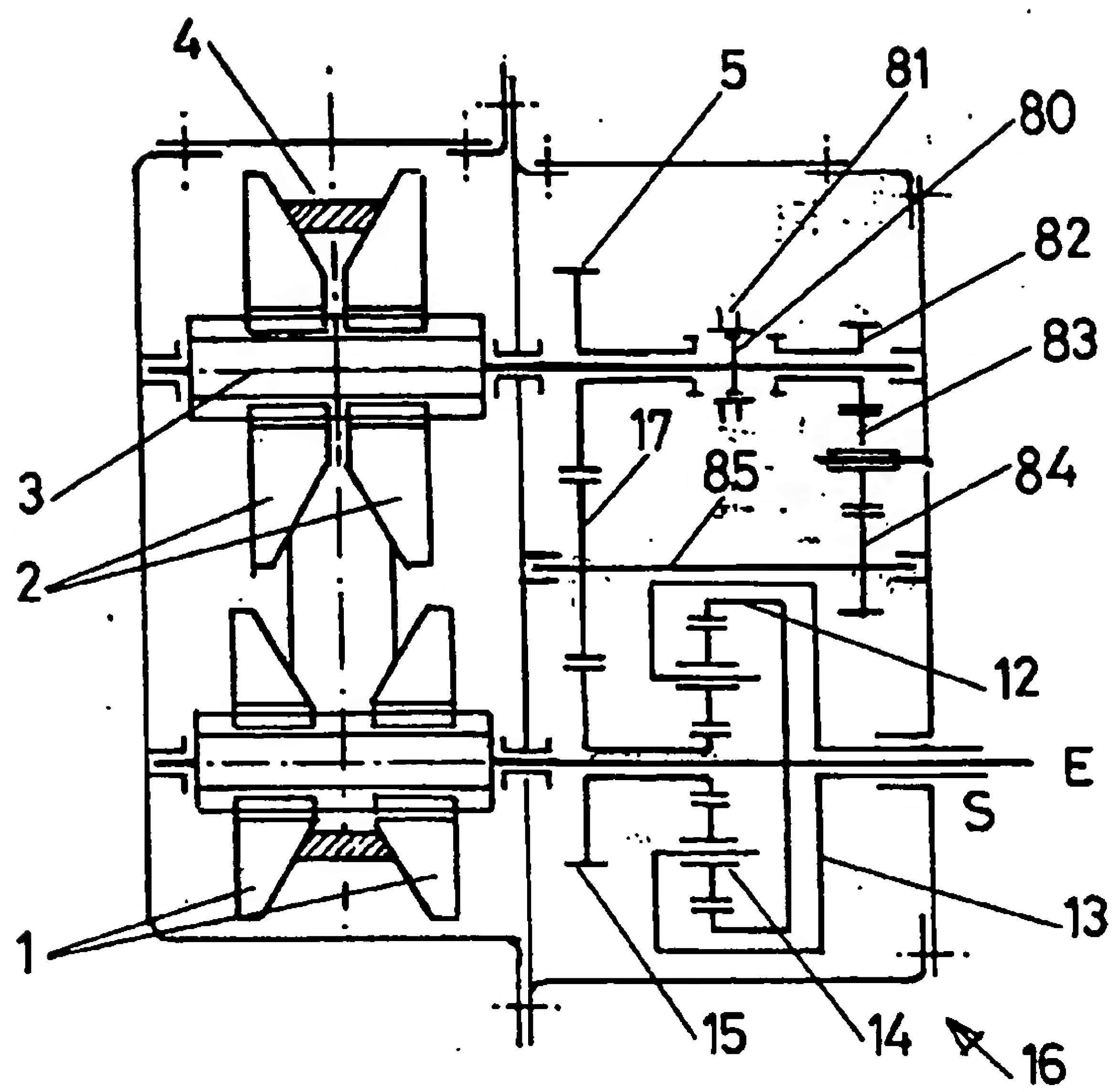

FIGURE 28